

INTAKE PORT

PROVISIONAL APPLICATION REFERENCE

This application relates to
Provisional Application Number 60/397,786 having a filing date of 07/23/2002.

UNITED STATES GOVERNMENT NOTICE

This invention was made with Government support under Contract No. ANL-2F-00501 awarded by the Department of Energy. The Government has certain rights in this invention.

BACKGROUND OF THE INVENTION

Prior art intake ports are typically tuned in length and in cross section area to provide either increased torque at a design speed, or a more uniform torque curve through a range of engine speeds. It is common for prior art ports to have a funneled shape, with a minimum cross section area immediately upstream of the intake valve seat(s). The port length provides resonant tunings and the small port cross-section area at the outlet of the port acts as a nozzle for providing a high outlet flow velocity. The intake port is defined as the intake flow path from the intake plenum extending downstream to the intake valves.

The ratio of minimum port cross section area just up stream of the intake valve seat(s) to port cross section area at the inlet to the cylinder head (and outlet of the manifold) is typically about 1.0. A smaller port cross section area at the cylinder head inlet and extending down to the valves provides a faster flow velocity due to more nozzling of the flow, but the flow coefficient or port Cd value is generally compromised. Some ports have relatively large cross section area flowing into the cylinder head for more aerodynamic flow and a larger flow coefficient value Cd, however these ports tend to have a lower flow exit velocity due to less nozzling of the flow. The increase in cross section area improves the flow coefficient through the valves at the expense of the ram tuned exit velocity.

Venturi gaskets have been sold in the aftermarket by Spoon Sports in Japan and their distributors in the United States that significantly reduce port cross section area at the cylinder head inlet. These venturi gaskets can modestly increase engine torque in engines that do not have very good port flow to begin with. In a typical application, torque is modestly improved in an area where torque is low to begin with, and maximum torque is no better or not significantly better than a well-designed conventional port. The venturi gaskets provide very little benefit for engines that have good port flow to begin with. With these venturi gaskets, the volume inside of the port down stream of the intake manifold is relatively small. The venturi is located between the intake manifold and cylinder head a short distance from the intake valves, and consequently the divergent nozzle is short and abrupt. The short nozzle length causes a sudden expansion of the flow and inefficient diffusion and pressure recovery.

SUMMARY OF THE INVENTION

According to the present invention, the intake port of a reciprocating piston internal combustion engine includes a venturi located almost halfway upstream the intake port between the intake valves and intake plenum, for providing a high flow velocity ram effect; a large lower port head volume for providing highly aerodynamic flow through the intake valve(s) into the working cylinder; and a high efficiency diffuser for converting the high speed flow from the venturi into elevated gas pressure with minimal losses.

The very small venturi throat or nozzle area plus tuning of the port upstream of the venturi provides an exceptionally high flow velocity through the nozzle. The long diffuser downstream of the venturi efficiently converts the high-speed flow into elevated gas pressure. The large lower head volume holds the pressurized intake air for blow down through the intake valve(s) into the working cylinder. The large cross section area of the head also enables an exceptionally high port flow coefficient C_d to be attained. The venturi is located far upstream to provide a large head volume plus port length needed for providing a high efficiency diffuser. Placing the venturi far upstream permits the venturi nozzle area to be exceptionally small and in turn the ram velocity exceptionally high. The combination of exceptionally high ram velocity, high efficiency

diffusion, large head volume, and high flow coefficient C_d values made possible by the large head cross section area provides an exceptionally high volumetric efficiency and in turn exceptionally high engine torque levels. Torque is significantly improved throughout the engine's full operating speed range in most embodiments of the present invention.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is intended to illustrate a sectional view of an intake port according to the present invention.

Fig. 2 is a chart showing intake port cross-section area vs. port length according to the present invention.

Fig. 3 is similar to Fig. 1 and provides greater detail on the specifications of the port according to the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Fig. 1 is intended to illustrate an intake port 1 according to the present invention having an intake plenum 2, an inlet funnel 3 and a port inlet 4, a ram tuning pipe 5, a ram nozzle 7 having a venturi throat 9, a diffuser 10, a surge chamber 12, and an outlet nozzle 14, having an inner seat diameter or ISD 16 at its outlet. The port further includes a flow centerline 18. The port is preferably attached to the combustion chamber and working cylinder of a 4-stroke internal combustion engine 22, having one or more intake valves 20 per cylinder and one or more exhaust valves per cylinder 24 per cylinder mounted in a cylinder head 23. A piston 21 is mounted in cylinder 22 for reciprocating motion within the cylinder. The reciprocating piston in the cylinder defines a swept cylinder displacement D_c equal to the cross section area of cylinder 22 times the stroke of the piston. The intake port may optionally include one or more fuel injectors 26.

The intake port is defined as the intake flow path from the intake plenum extending downstream to the intake valves. Technical literature commonly refers to the intake port as the part of the flow path situated in the cylinder head, and the intake runner as the part of the flow path situated in the intake manifold that extends from the intake plenum downstream to the cylinder head inlet. This prior definition is based on the intake manifold to cylinder head parting line, which has an arbitrary location relative to the gas path aerodynamics. This common definition of port is not used in the present specification, and instead the intake port is defined as the full intake flow path from intake plenum to intake valves.

Fig. 2 is intended to illustrate the cross section area of the port through its length, where port length is shown on the horizontal axis of Fig. 2, and port cross section area is shown on the vertical axis of Fig. 2 for an intake port according to the present invention. The port section reference numbers described with relation to Fig. 1 are shown in Fig. 2.

Fig. 3 is a detailed view of an intake port according to the present invention. Inner seat diameter 16 is the smallest diameter that intake valve 20 comes into contact with. Each inner seat diameter 16 has an inner valve seat cross-section area S_a . Intake port 1 has a total outlet area 17 equal to the sum of the inner valve seat cross section areas S_a for all of the intake valves 20 in fluid communication with intake port 1. Intake port 1 has an equivalent outlet diameter 19 being equal to the square root of the quantity total outlet area 17 times four divided by pi,

$$\text{Equivalent outlet diameter} = (4 \times \text{total outlet area} / 3.14159)^{1/2}$$

Intake port 1 has an overall port length L_p measured along flow centerline 18 from inner valve seat cross-section area S_a to port inlet 4. In embodiments of the present invention having more than one intake valve per cylinder, the overall port length may be longer for one valve, and in this case the overall port length L_p for the port is defined as the average overall port length for all of the valves in the port.

Intake port 1 has a tuning length L_t measured along flow centerline 18 from inner valve seat cross section area S_a to venturi throat 9 also referred to as V_t . Venturi throat V_t is the cross section of the port having the smallest cross section area of the port,

excluding the port leg cross section areas near inner valve seat cross section area Sa. The minimum cross section area may extend for a length Lm of the port, and in this condition venturi throat Vt is assumed to be located at the mid point of length Lm.

In embodiments of the present invention having more than one intake valve per cylinder, the tuning length may be longer for one valve, and in this case the tuning length Lt for the port is defined as the average tuning length for all of the valves in the port.

Venturi throat Vt has a venturi cross-section area Va and an equivalent diameter Vd, Vd being equal to

$$Vd = (4Va/\pi)^{1/2}$$

The venturi further has characteristic length or venturi length Lu having an inlet end and an outlet end, the venturi inlet end being located a distance upstream of venturi throat Vt equal to 5 percent of the tuning length Lt, and the venturi outlet end being located a distance downstream of venturi throat Vt equal to 5 percent of the tuning length Lt. In more detail, venturi length Lu is 10% of the tuning length Lt, and the mid point of venturi length Lu is centered on venturi throat Vt so that half of venturi length Lu extends upstream of venturi throat Vt and half of venturi length Lu extends down stream of venturi throat Vt. The venturi further has a characteristic volume or venturi volume Vv defined as the volume of the port within the venturi length Lu, and in more detail within the cross sections located at both ends of venturi length Lu, the cross sections being perpendicular to the centerline of the port flow stream centerline 18.

Intake port 1 further has a surge chamber 68 having a surge chamber inlet end 70 and a surge chamber outlet end 72, the surge chamber inlet end 70 being located a distance downstream of venturi throat Vt equal to 30 percent of the tuning length Lt, that is the surge chamber inlet 70 is located 30 percent of the tuning length Lt downstream of venturi throat Vt, and the surge chamber outlet end 72 is located a distance upstream of inner valve seat cross section area Sa equal to 20 percent of the tuning length Lt, that is surger chamber outlet 72 is located 20 percent of the tuning length Lt upstream of inner valve seat cross section area Sa. Surge chamber 68 has a surge chamber volume within intake port 1 and between surge chamber inlet end 70 and surge chamber outlet end 72.

Intake port 1 has a ratio of venturi cross-section area V_a to total outlet area 17. In the preferred embodiment of the present invention, the ratio of venturi cross-section area V_a to total outlet area 17 is no more than 0.87. Additionally, according to the present invention the volume of surge chamber 68 is greater than the product of total outlet area 17 times 0.26 times tuning length L_t . Additionally, according to the present invention, venturi volume V_v is less than the product of total outlet area times 0.09 times the tuning length L_t . Accordingly, the port has a large surge chamber and a small venturi for providing a high volumetric efficiency for the engine.

Several parameters are required to define the intake port according to the present invention. Some or all of the variables may be optimized within the spirit and scope of the present invention. Preferably tuning length L_t is greater than 115 millimeters. A tuning length of 115 millimeters provides generally good performance for a broad range of automotive size engines. Intake port 1 further has a ratio of tuning length L_t to equivalent outlet diameter 19. Preferably the ratio of tuning length L_t to equivalent outlet diameter 19 is at least 2.4 in order to provide good performance for a broad range of automotive size engines. Intake port 1 further has a ratio of surge chamber volume 68 to venturi volume V_v . Preferably the ratio of surge chamber volume 68 to venturi volume V_v is at least 4.6 but no more than 8.0 in order to provide a large surge chamber volume, but not too large a surge chamber volume, as too large a volume can cause the flow to stall in some embodiments of the present invention. Intake port 1 further has a ratio of tuning length L_t to overall port length L_p . Preferably the ratio of tuning length L_t to overall port length L_p is at least 0.325. Intake port 1 further has a ratio of venturi cross-section area V_a to cylinder displacement D_c . Preferably the ratio of venturi cross section area to cylinder displacement is no more than 0.0022 square millimeters of venturi cross section area per cubic millimeter of cylinder displacement in order to provide a small venturi cross section area for high speed flow and a large surge chamber volume for blow down of intake air or another fluid into the working cylinder of the engine.

The present invention provides an exceptionally high degree of diffusion between the venturi throat V_t and surge chamber volume 68. Fuel injector 26 injects fuel into port 1 through an injector pocket 27. Injector pocket 27 can disrupt diffusion of the flow between venturi throat V_t and surge chamber volume 68. According to the present

invention, venturi throat V_t is moved upstream and/or injector pocket 27 is moved downstream so that a large degree of diffusion can be completed before the injector pocket disrupts the flow. Fuel injector 26 includes an injector tip 26t and intake port 1 includes an injector tip port section 74 that passes through injector tip 26t, injector tip port section 74 having an injector port cross section area 76. Preferably, according to the present invention, the injector tip port cross section area 76 is greater than the sum of 0.75 times venturi cross section area V_a plus 0.25 times total outlet area 17.

As stated earlier, some or all of the port specifications may be optimized within the spirit and scope of the present invention to provide a high performance port for a broad range of automotive applications. Preferably, according to the present invention, the ratio of venturi cross section area V_a to total outlet area 17 is no more than 0.87; the volume of surge chamber 68 is greater than the product of total outlet area 17 times 0.26 times tuning length L_t ; venturi volume V_v is less than the product of total outlet area times 0.09 times the tuning length L_t ; tuning length L_t is greater than 115 millimeters; the ratio of surge chamber volume 68 to venturi volume V_v is at least 4.6; and the ratio of venturi cross section area to cylinder displacement is no more than 0.0022 square millimeters of venturi cross section area per cubic millimeter of cylinder displacement. Accordingly, the port provides a high flow velocity ram effect, a large lower port head volume for providing highly aerodynamic flow through the intake valve(s) into the working cylinder, and a high efficiency diffuser for converting the high-speed flow from the venturi into elevated gas pressure with minimal losses. In some embodiments of the present invention port flow can be further improved with a ratio of venturi cross section area V_a to total outlet area 17 less than 0.68; plus a surge chamber volume 68 greater than the product of total outlet area 17 times 0.31 times the tuning length L_t ; plus a venturi volume V_v less than the product of total outlet area 17 times 0.08 times the tuning length L_t ; plus a ratio of surge chamber volume 68 to venturi volume V_v of at least 5.3. In some embodiments of the present invention, this specification can be further improved by reducing the ratio of venturi cross section area to cylinder displacement to no more than 0.0021 square millimeters of venturi cross section area per cubic millimeter of cylinder displacement, and/or increasing the tuning length L_t to at least 135 millimeters. With an increased tuning length, the injector tip port cross section area is preferably

greater than the sum of 0.75 times venturi cross section area V_a plus 0.25 times total outlet area 17, and the ratio of tuning length L_t to equivalent outlet diameter 19 increased to at least 3.1. In some applications of the present invention, the tuning length L_t may be further increased to 145mm and or the ratio of venturi cross section area V_a to total outlet area 17 reduced to no more than 0.625, thereby providing an exceptionally small venturi, a large diffuser, and a large surge chamber volume.

According to the present invention, an exceptionally high flow velocity through the venturi is achieved, followed by an exceptionally large amount of diffusion as the flow travels into the surge chamber volume. The flow then blows down from the large surge chamber through the intake valves into the working cylinder of the engine, providing an exceptionally high volumetric efficiency and high engine power output.

The present invention may be practiced within the spirit and scope of the claims. The present specifications have been developed for engines having two intake valves per cylinder, but are expected to also provide significant performance improvements for engines having one or three intake valves per cylinder. Additionally, the specifications have been developed for engines having an individual cylinder displacement less than 600 and more than 400 cubic centimeters or cc, or 0.6 and 0.4 liters, but are expected to also provide significant performance improvements for engines having a larger or smaller displacement. Additionally, the specifications have been developed for engines having a maximum speed less than 7000 rpm, but are expected to also provide significant benefits for engines operating at higher speeds.